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# Study of dual-fuel (diesel + natural gas) particle matter and $CO_2$ emissions of a heavy-duty diesel engine during transient operation

*The aim of this study is to describe the impact on particle matter and $CO_2$ emissions of converting an existing heavy-duty diesel engine for on-highway truck applications to a dual-fuel engine (diesel + natural gas), especially in transient operation. A dual-fuel engine with homogeneous gas charge injection in the intake line before turbocharger was considered. The results showed the feasibility of this kind of technology for transient operation reaching a significant reduction of particle matter plus a decrement in $CO_2$ emissions at the expense of a small decrement of brake fuel conversion efficiency and an increment of unburned hydrocarbons in the exhaust gases.*

Key words: *dual-fuel, natural gas, diesel engine, heavy-duty, transient, particle matter, $CO_2$*

## 1. Introduction

Nowadays there is significant interest in converting diesel to NG for heavy-duty engines used in commercial vehicle applications due to the growing availability of NG in Europe which opens the way to using it in long-distance transport. This also reduces Europe's dependence on liquid oil fuels and the risk for the future stability of supply resulting from a significant price increase. Another important advantage is that the use of NG can be favorable to decreasing or mitigating $CO_2$ emissions and has potential for reducing toxic exhaust specific emissions such as smoke and PM.

The mentioned diesel to NG engine conversion can be total or partially performed. In the case of total conversion, the combustion system is changed in order to work from diesel to otto-cycle; this conversion can include several modifications such as changing the combustion chamber in order to obtain a new compression ratio and introducing spark plugs. The technical process and the main technological limits that apply to existing European diesel engine for truck application currently used in long-distance transport converted to a dedicated NG engine is widely studied in references [1, 2].

In the case of partial engine conversion heavy-duty dual fuel (HDDF) engine is used [3–7]. A HDDF engine is conceived to simultaneously operate with diesel fuel and a gaseous fuel, both fuels being metered separately, where the consumed amount of one of the fuels relative to the other one may vary depending on the operation. Two types of technology are considered according to the way of gas injection: High Pressure Direct Injection (HPDI) and the Homogeneous Gas Charge Injection (HGCI). In the first one diesel and NG are injected directly into the combustion chamber at higher injection pressure using a special injector with a dual-concentric needle design or by using two separate injectors [8, 9]. This allows for small pilot quantities of diesel fuel and large quantities of NG to be delivered at high pressure to the combustion chamber. The major advantages of this type of HDDF engines are that there is no limitation by knocking at high loads, their HC emissions are low and they can replace more than 90 % of the diesel fuel (by energy). The main disadvantages for this kind of technology are the introduction of the new injection system which can lead to some important engine cylinder head modifications for retrofitted engines, a direct communication with OEM controller software could be necessary and that such engines only work with LNG, which means a cryogenic system should be installed. This technology is outside the scope of this work.

In the case of HGCI, gaseous fuel is injected into the intake line of the engine (Fig. 1) and premixed with air or EGR during the intake and compression stroke. Ignition of the charge is managed by injection and auto ignition of diesel fuel using the original injection system of the baseline engine. Substitution rates of 50 % to 75 % can be achieved depending on several factors including duty cycle and the level of integration with the base diesel. The most important advantages of this technology are the simplified engine control with no direct communication with OEM, the non-intrusive technical modifications in the engine structure, the fact that it can be used with CNG and LNG indistinctively, which is a key factor in places where gas infrastructure and availability play an important role, and the low cost for the implementation in retrofitted engines with a considerable time reduction during development phase comparing to HPDI. The main disadvantages of HGCI technology are that the upper load range is typically limited by knocking, higher CO and HC emissions which lead to using methane catalyst in the exhaust pipeline and that the transient response of the engine could deteriorate as substitution rates increase especially during transient operation.

Bearing in mind the benefits and drawbacks for HDDF engine operation the objective of this work is established. The aim of the present work is to study the impact in PM and $CO_2$ emissions of converting existing HD diesel engine for on-highway truck applications to a HDDF engine when Homogeneous Gas Charge Injection in the intake line before turbocharger is used.

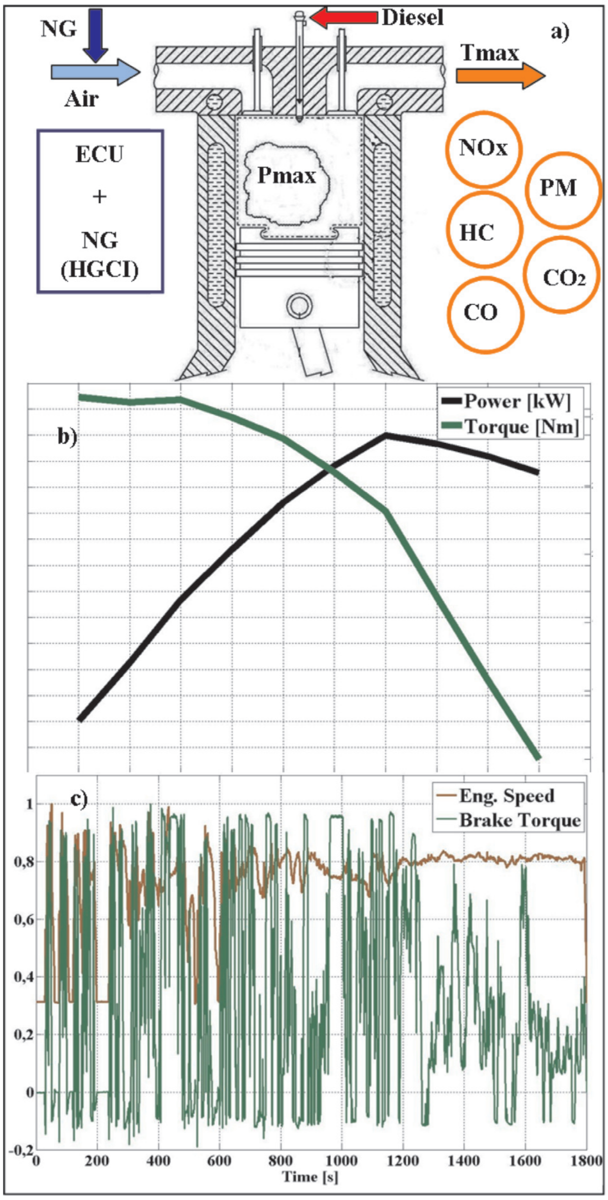



Fig. 1. HDDF engine concept (HGCI): a) emissions and thermodynamics, b) performance, c) transient behaviour

## 2. Experimental set-up

The experimental tests were carried out in a turbocharged intercooled heavy-duty DI diesel engine for on-highway truck applications. The engine used in this work was a four-stroke, six cylinder diesel engine equipped with an EUI fuel injection system. Table 1 summarizes the main engine specifications. The engine was fully instrumented with fast response thermocouples and mean pressure sensors, which were placed in convenient zones. The temperature of all the fluids: inlet air, cooling water, lube oil, EGR and fuels were measured by PT100 and K-type thermocouples. In addition, in order to obtain the maximum in-cylinder pressure and the maximum inlet turbocharger turbine during diesel operation AVL Indimodul 621 indicating system and the exhaust manifold fully instrumented with temperature and pressure sensors were used respectively.

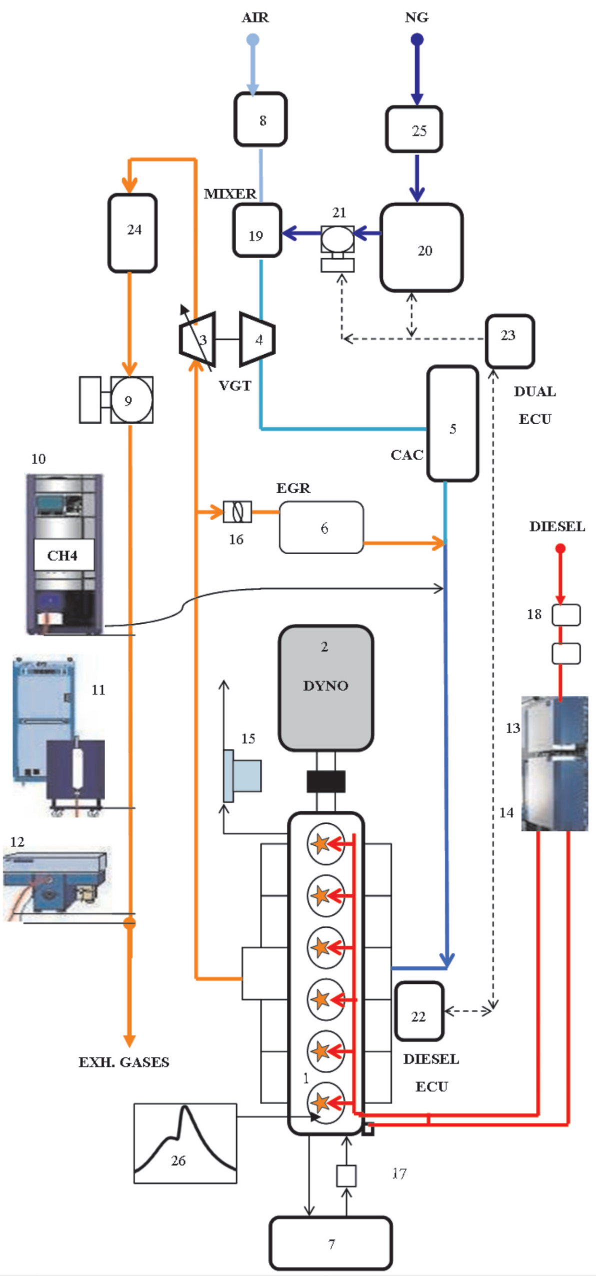
Table 1. Main engine characteristics

| Type | Turbocharged – intercooled DI Diesel engine with Variable Geometric Turbine (VGT) and High Pressure cooled EGR |
|---|---|
| No Valves | 4 each cylinder |
| No Cylinders | 6 in-Line |
| Bore x Stroke | 134 x 168 mm |
| Compression Ratio | 16.8:1 |
| Injection System | Electronic Unit Injector (EUI) |
| Power level | >340 kW @ 1800 rpm |
| Torque level | > 2000 Nm @ 1200 - 1500 rpm |

The test cell was equipped with an active dynamometer with correspondingly maximum values of: power (450 kW), torque (2869 N·m) and speed (4500 rpm). The dynamometer is equipped with digital high-precision torque transducer HBM T12 and is linked with a PC via AVL PUMA OPEN 1.3 software to program and run the stationary and transient tests automatically. A scheme of the experimental set-up employed in this work is shown in Fig. 2.

Table 2. Main properties of the working fuels

| Diesel properties | | |
|---|---|---|
| Cetane number | 52.8 | [-] |
| Density @ 15° C | 836 | kg/m$^3$ |
| Viscosity @ 40° C | 2.891 | mm$^2$/s |
| Sulphur | 3.0 | mg/kg |
| Carbon | 85.91 | % weight |
| Hydrogen | 13.43 | % weight |
| LHV | 42.917 | MJ/kg |
| **NG properties** | | |
| Methane | 92.0347 | %Mole |
| Ethane | 7.0087 | %Mole |
| Propane | 0.6620 | %Mole |
| Butane | 0.1052 | %Mole |
| Pentane | 0.0020 | %Mole |
| Iso butane | 0.0754 | %Mole |
| Iso pentane | 0.0073 | %Mole |
| Nitrogen | 0.1047 | %Mole |
| Carbon dioxide | 0.0000 | %Mole |
| LHV | 10.5952 | kWh/m$^3$ |
| Density | 0.7743 | Kg/m3 |




| | | | |
|---|---|---|---|
| 1 | Diesel engine | 14 | AVL 753C fuel temp. control |
| 2 | Active dynamometer | 15 | AVL 442 blow-by meter |
| 3 | Turbocharger turbine (VGT) | 16 | EGR control valve |
| 4 | Turbocharger compressor | 17 | Coolant flow meter |
| 5 | Charge air cooler (CAC) | 18 | Fuel filters |
| 6 | EGR cooler | 19 | NG and air mixer |
| 7 | Engine coolant cooler | 20 | NG press regulator system |
| 8 | Air mass flow meter | 21 | NG throttle valve |
| 9 | Exhaust back pressure valve | 22 | Diesel ECU |
| 10 | Horiba 7500 DEGR + CH4 module | 23 | Dual ECU |
| 11 | Horiba MDLT 1300 T | 24 | Methane catalyst |
| 12 | AVL 439 opacimeter | 25 | NG measurement device |
| 13 | AVL 735S fuel mass flow meter | 26 | AVL indimodule |

Fig. 2. General schema of the engine installation at the test cell

The test cell can work with different types of fuels such as gasoline RON 95/98, diesel, NG, LPG, reference and special fuels (leaded gasoline for example). Major characteristics of the diesel and NG fuels used in this work are shown in Table 2. Diesel fuel consumption was measured by AVL 735S continuous measurement system (Coriolis principle) and diesel fuel temperature control was achieved by AVL 753C equipment.

NG fuel consumption was measured by a BRONKHORST gas mass flow meter with a measurement range of 1–150 kg/h. The main parts of the HGCI system are: an air-NG mixer, a gas throttle electric valve and the NG pressure regulator system (a shut-off electric valve, a high pressure limiter and a low pressure regulator). These components are continuously communicated with dual ECU.

The intake air consumption was measured by Sensyflow P thermal mass flow meter with a measurement range of 0–4000 kg/h. The test cell is also equipped with humidity and intake air temperature control system.

For the measurement of the typical gaseous emissions ($NO_x$, $CO_2$, CO, HC) exhaust gas analyzer HORIBA 7500DEGR was used which was extended with a $CH_4$ module for its measurement during dual-fuel mode. It is important to mention that methane catalyst was only installed after all the diesel baseline tests were done.

To construct the 1D model for diesel baseline engine which is explained in the next section, $CO_2$ in the intake manifold was also measured to calculate EGR rate according to equation (1) where the ambient $[CO_2]_{atm}$ concentration remains constant at 0.04 %.

$$EGR = \frac{[CO_2]_{intake} - [CO_2]_{atm}}{[CO_2]_{exhaust} - [CO_2]_{atm}} \qquad (1)$$

The PM specific emission during standard emission cycles was measured by micro dilution tunnel HORIBA MDLT-1300T. However, for HDDF engine calibration tasks in the test bench a partial flow Opacimeter AVL 439 was used to measure the smoke opacity of a sample of the exhaust flow. The opacity values obtained from diesel baseline engine were used as a reference.

HDDF engine operation was controlled by means of IDIADA engineered dual ECU. The management of the Diesel quantity by controlling diesel ECU and the gaseous injected quantity by controlling gas throttle electric valve operation is done by this controller. This ECU also performs monitoring tasks during diesel and dual-fuel engine operation through CAN bus J1939. The Dual ECU was also tested in an automatic HiL test bench for simulation of the hardware environment and to put it through all the possible scenarios it might experience in real world environment. Several tests according to ISO16750 were also performed.

## 3. Methodology

To reach the objective of the present work two major activities were planned as stated in Table 3: the first of them was to perform dual-fuel engine calibration which comprises a virtual (1D model) and test bench calibration in order to have the final calibration maps for the dual

ECU. The second major activity was to develop worldwide standard emission cycles especially in transient operation to check engine behavior and compare brake fuel conversion efficiency, energy distribution and exhaust specific emissions values with those previously obtained in diesel fuel operation only.

### 3.1. Calibration activities

The main target of virtual calibration based on 1D simulation model was to obtain a baseline calibration for the engine in dual-fuel mode for a quicker and less expensive development process. It is important to mention that this is a feedback loop process as can be seen in Fig. 3.

First, the 1D model was constructed using GT-power software which is the industry-standard engine simulation tool. This tool takes account of wave dynamic manifold phenomena via robust solution of the Navier-Stokes equations and the transient performance response of the engine that could be an important issue when HGCI technology is used. After that, the model was extensively validated in diesel mode with experimental data from the engine test bench.

Once the model was validated the next step was to simulate and optimize strategies and control devices functioning in dual-fuel mode. In this work due to the engine original technology (EGR + VGT) the EGR deactivation strategy when the engine works in dual-fuel mode was selected in order to have a simple electronic control by dual ECU. This strategy was extensively tested in the 1D model before it was finally on the test bench.

Table 3. Experimental matrix

| Tasks | Subtasks | Dual-fuel Mode | Target |
|---|---|---|---|
| Calibration tasks | Virtual calibration (1D model) | Engine mapping (no idle) | Fast engine development<br>EGR deactivation<br>Gas throttle control<br>Baseline calibration |
| | Test bench calibration | | Diesel replacement ratio<br>Stationary calibration<br>Transient adjustments<br>Final calibration maps |
| Emission cycles | ESC | 9 modes (25-75% Load) | Dual transitions<br>PM |
| | FTP | Crowded freeway (300 s) | Transient behaviour<br>$CO_2$ & PM<br>Brake fuel conversion efficiency |
| | ETC | Motorway (300 s) | Transient behaviour<br>Fuel energy distribution |

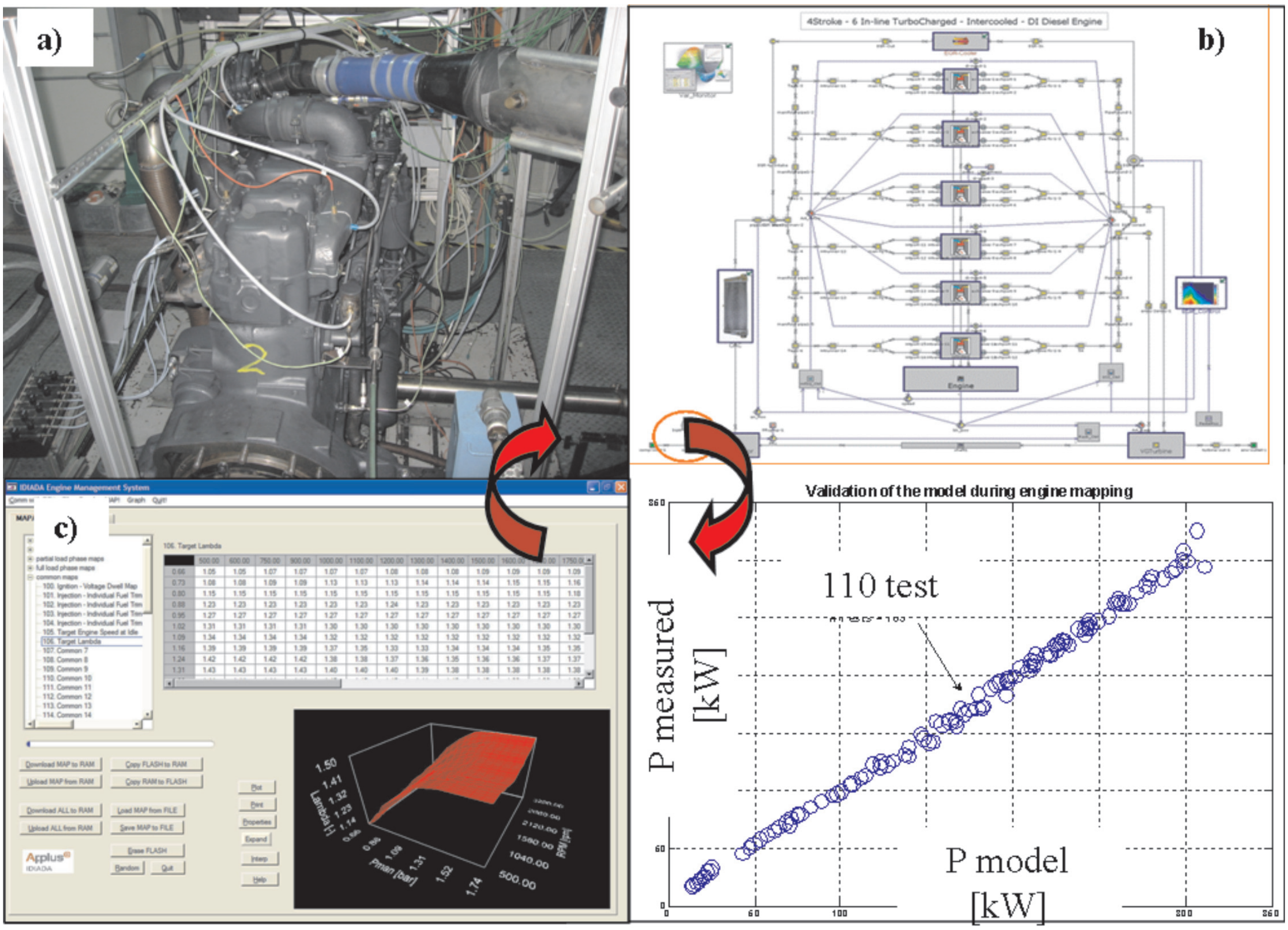



Fig. 3. General methodology for engine calibration: a) testing, b) modelling and validation, c) baseline calibration

Some other important targets reached by means of the 1D engine model were to infer the engine dynamic behavior in dual-fuel mode testing several substitution rates via Simulink coupling (Fig. 4a) and to select an optimized gas throttle electric valve diameter (Fig 4b). The response of this valve was tested in different stationary and transient scenarios in the model or even in extreme conditions such as when the engine is suddenly forced to leave dual-fuel mode to diesel operation or vice versa.

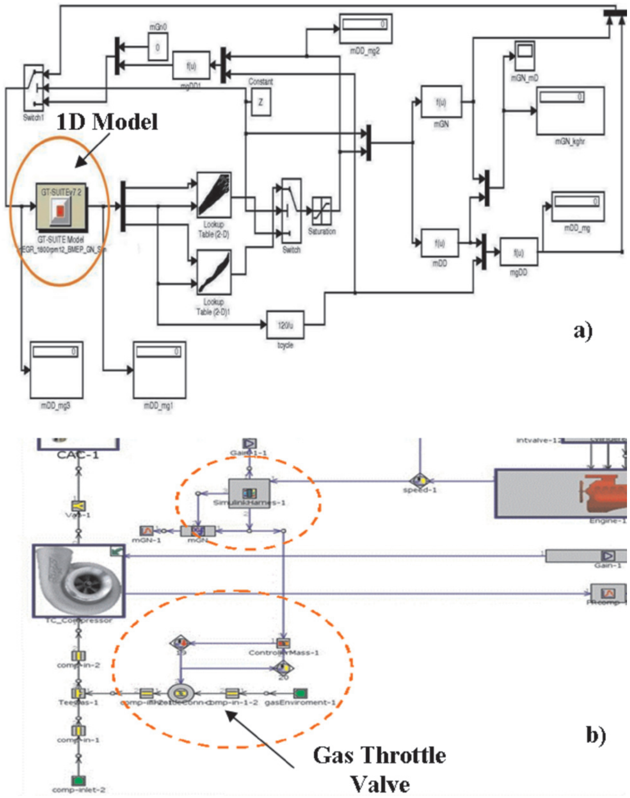



Fig. 4. Modelling dual-fuel mode operation: a) dynamic control with Simulink, b) natural gas throttle valve operation

On the other hand, the test bench calibration activities were done based on the previous baseline calibration from engine model. This baseline calibration was downloaded to dual ECU (Fig. 5) and each point was slightly and conveniently modified at the test bench to match engine performance and emissions.

The stationary calibration was performed in order to reach the maximum Gas Energy Ratio (GER) until the engine's thermodynamic limits (Fig. 6a). Examples of these restrictions were maximum in-cylinder pressure (Fig. 6b), maximum rate of pressure rise and maximum temperature measured at inlet turbocharger turbine. Final dual-fuel mode calibration maps were achieved after the implementation of small adjustments via electronic gains and filters for engine transient operation. The calibration was broadly tested in steady state and transient conditions at the test bench to verify its functionality.

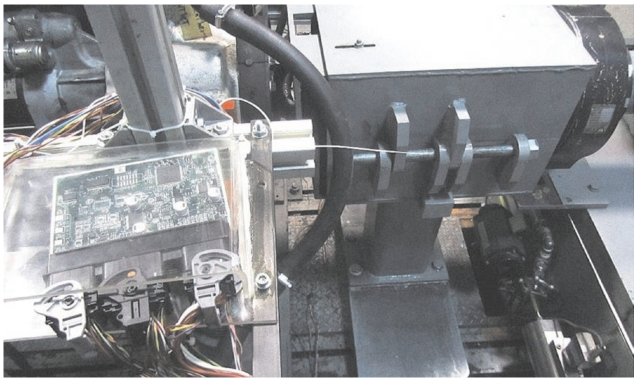


Fig. 5. IDIADA engineered dual ECU during test bench engine calibration

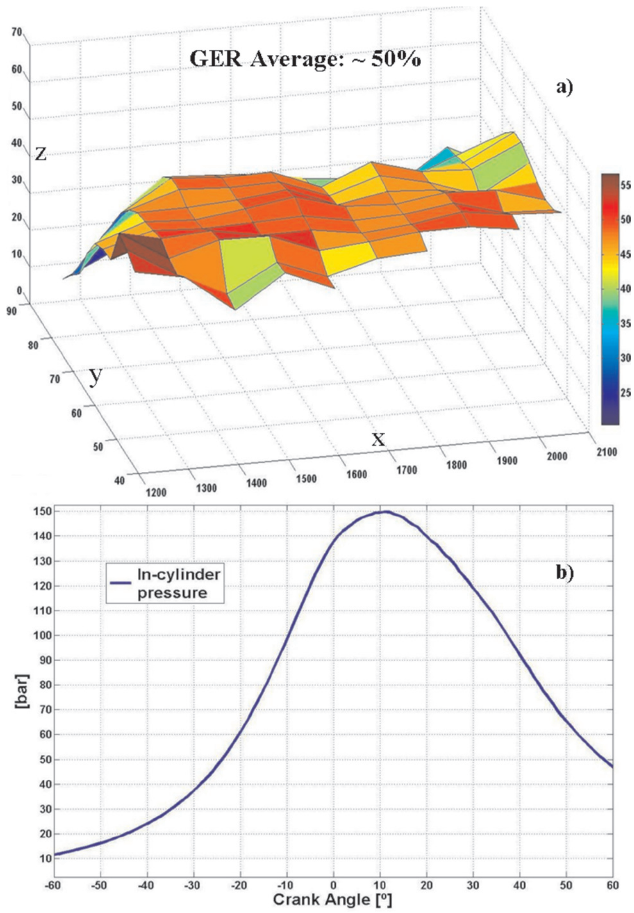



Fig. 6. Final calibration results: a) average GER, b) maximum in-cylinder pressure

Chemical energy supply, GER, and the brake fuel conversion efficiency in dual-fuel mode were calculated by equations (2), (3) and (4) respectively.

$$E_{Dual} = m_{NG} \cdot LHV_{NG} + m_{DD} \cdot LHV_{D} \tag{2}$$

$$GER = 100 \cdot \left( \frac{m_{NG} \cdot LHV_{NG}}{E_{Dual}} \right) \tag{3}$$

$$\eta_{Dual} = \frac{W_{Effective}}{E_{Dual}} \tag{4}$$

where: $m_{NG}$ is the mass of NG injected [kg], $m_{DD}$ is the mass of diesel fuel injected [kg], $LHV_D$ is the lower heating value of diesel [kJ/kg], $LHV_{NG}$ is the lower heating value of NG [kJ/kg], $W_{Effective}$ is the effective work by the engine [kJ], $E_{Dual}$ is chemical energy supply by dual-fuel [kJ], GER is the Gas Energy Ratio [%], $\eta_{Dual}$ is the brake fuel conversion efficiency [–].

It is important to mention that maximum GER that engine permitted was around 50 % due to combustion instabilities appeared when this ratio was increased in several speed-load conditions. This limit could be increased if the management of the start of diesel injection is implemented in the dual ECU.

### 3.2. Emission cycles

The Federal Test Procedure (FTP) from U.S. and the European Transient Cycle (ETC) were the two worldwide standard tests selected to check transient behavior and specific emissions during dual-fuel mode operation. The crowded freeway part of the FTP and the motorway part of the ETC were chosen because the authors consider that these speed-load conditions are representative of long-distance transport and in these scenarios the major advantages of dual-fuel mode operation such as emissions reduction and significant fuel energy replacement per distance travelled are achieved. Even though the motorway part of the ETC has a total duration of 600 s only the last 300 s were studied in dual-fuel mode in order to have the same baseline time duration of the FTP crowded freeway.

The European Stationary Cycle (ESC) was also tested to check the potential PM reduction in steady-state for this kind of technology. The full load condition of the three main speeds of the ESC test were selected to only work in diesel in order to check the engine behavior when diesel-dual transitions are presented at high loads; therefore, dual-fuel mode upper calibration limits were moved down from 100 % to 95 % of load. All results in the tests were compared with those obtained in the same conditions in diesel original operation (with EGR) and are presented in the result section as normalized values (diesel = 100 %).

The specific gaseous and PM emissions in [g/kWh] were calculated by means of in-house IDIADA software (Fig. 7) which is a GUI developed in Matlab environment. This tool has been extensively proved with certificated engines and it incorporates worldwide emissions standards libraries not only for HD and off-road vehicles but also for light-duty and passenger cars.

Diesel emission calculations were done according to the corresponding type approval engine model. In the case of dual-fuel, specific exhaust emission calculation was done using the procedure, definitions and recommendations pointed out in reference [10]. The mentioned document states that emission testing of a dual-fuel engine is complicated by the fact that the fuel used by the engine can vary between pure diesel fuel and a combination of mainly gaseous fuel with only a small amount of diesel fuel as an ignition source. The ratio between the fuels used by a dual-fuel engine can also change dynamically depending on the operating condition of the engine. As a result special precautions and restrictions are necessary to enable emission testing of these engines.

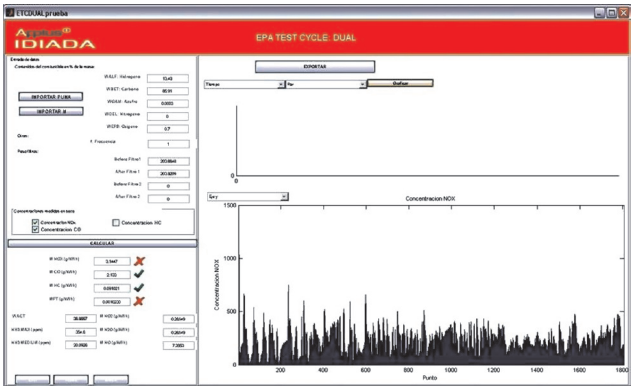



Fig. 7. In-house IDIADA emissions calculation software

## 4. Results

### 4.1 Stationary test: ESC

Figure 8a shows the nine modes chosen for dual-fuel mode during ESC test performed. The engine loads in these modes vary from 25 % to 75 % and the sum of their weight factors in the test represents 60 % of the total. The idle condition is always performed in diesel mode as was previously defined in Table 3. The results have confirmed the capability of the HGCI system to work in stationary operation and to perform diesel-dual transitions for example as changing from mode 2 in diesel (speed A at 100% load) to mode 3 in dual-fuel (speed B at 50 % load) without any problems for the engine.

The comparison of PM specific emission between the two ESC tests carried out in dual-fuel mode (9 modes in dual-fuel + 4 modes in Diesel) and the other only with diesel fuel is shown in Fig. 8b. The figure demonstrates the great potential of this kind of technology to reduce PM emissions reaching 62.5 % of reduction compared with diesel baseline engine.

### 4.2 Transient test: FTP

The FTP heavy-duty transient cycle comprises two main parts: a cold start and a hot start. The hot start is done after a stop of 1200 s completion of the cold start. Each two parts has four phases of 300 s duration each: the first is NYNF (New York Non-Freeway), the second is LANF (Los Angeles Non-Freeway), the third is LAFY (Los Angeles Freeway – crowded) and the fourth is a repetition of the first one. It can be seen from Fig. 9a that LAFY was the part selected to test the engine in dual-fuel mode. The statistical regression coefficient (R2) was higher than 95 % comparing engine torque in diesel and dual-fuel mode.

Figure 9b shows the comparison of the $CO_2$ emission evolution during the hot part of the LAFY. It can be seen from the figure that instantaneous $CO_2$ emissions in dual-fuel mode are lower than $CO_2$ emissions of the engine working only with diesel fuel in transient operation which demonstrates the potential of this kind of technology to reduce this type of GHG emission and therefore to contribute to decrease the total carbon footprint produced by the engine calculated as the total sets of its GHG emissions. Hence $CH_4$ emissions must be under control by methane catalyst.

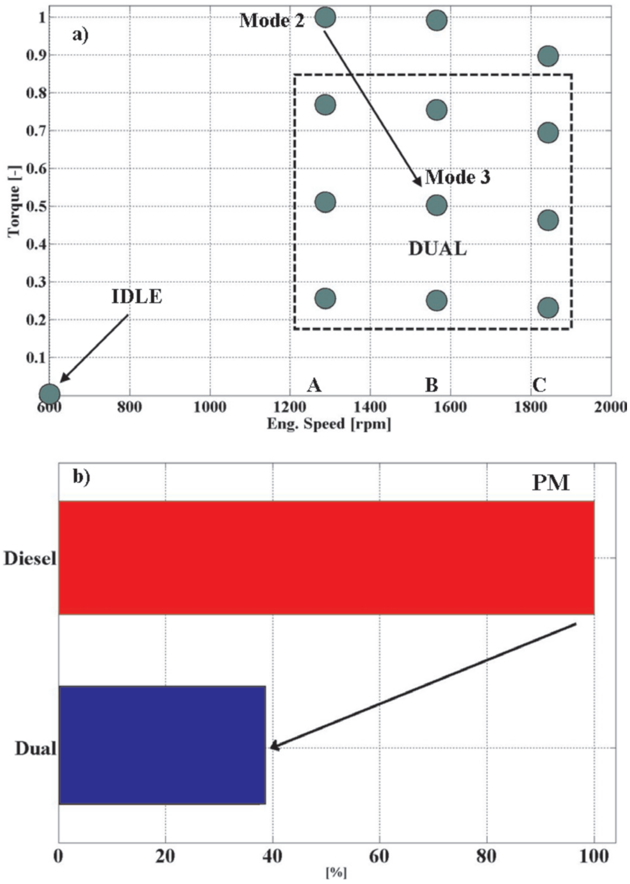



Fig. 8. ESC test: a) dual-fuel modes, b) PM comparison

Figure 10a shows the comparison of the results of $NO_x$, PM and $CO_2$ obtained after performing a complete sequence of FTP (cold + hot start) in diesel and dual-fuel mode. It can be seen from the figure that PM and $CO_2$ specific emissions decreased in dual-fuel mode 33.04 % and 3.64 % respectively. The reduction in $CO_2$ might have occurred because methane has very low C/H ratio which produces low $CO_2$ emission per energy content and the decrement in PM could be because of the short carbon chains present in the methane. On the other hand $NO_x$ specific emissions increased 20.01 % in the test. Even though nitrogen content in the NG (Table 2) could have contributed to increase $NO_x$ emissions, the authors consider that EGR deactivation strategy in dual-fuel mode might be the most influential factor for the result. This behavior is not expected in non-EGR diesel engines as it is reported in references [7, 11].

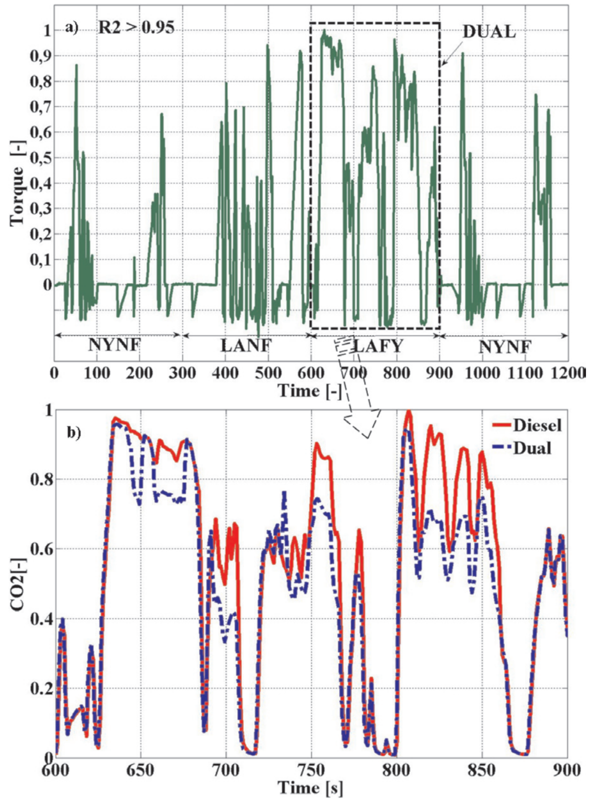



Fig. 9. FTP result: a) dual-fuel mode (crowded freeway), b) $CO_2$ comparison

Figure 10b shows the comparison of the results of CO, NMHC, and brake fuel conversion efficiency in the same condition mentioned in the above paragraph. The CO specific emissions decreased by 60.46 % but the NMHC increased by more than 100 %. The reduction of the CO was mainly due to methane catalyst. The relevant increment in NMHC could be improved by the implementation of the start of

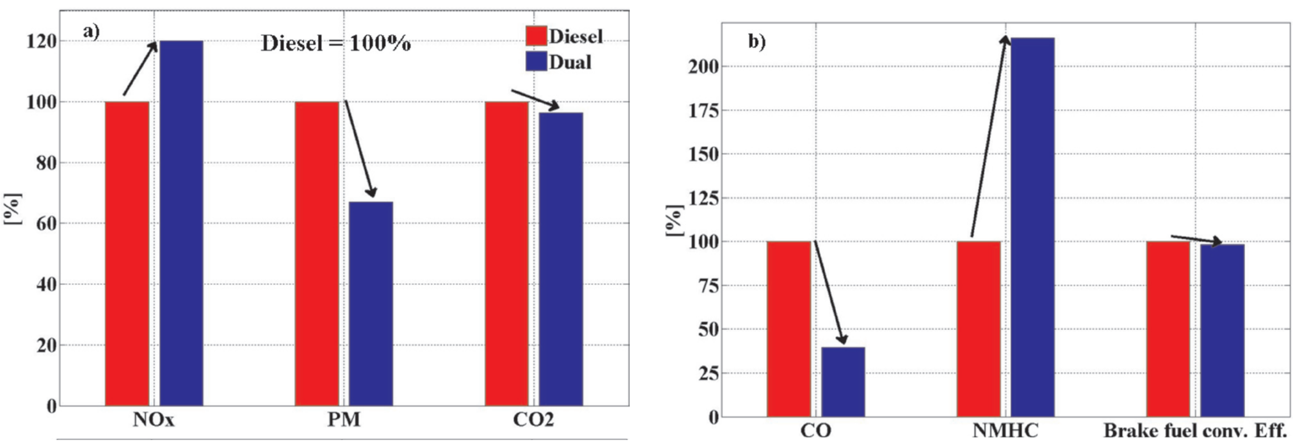



Fig. 10. FTP results comparison: a) $NO_x$, PM and $CO_2$ emissions, b) CO, NMHC emissions and fuel conversion efficiency

diesel injection management in the dual ECU or selecting a better methane catalyst with a higher HC efficiency conversion for working in transient operation. On the other hand, the brake fuel conversion efficiency only suffered a small decrement of 1.69 % during the test.

### 4.3 Transient test: ETC

The transient operation of the engine working in dual-fuel mode during the last 300 s of the motorway part from ETC test is shown in Fig. 11a (dotted line). In that condition engine speed was around 1600 rpm, meanwhile engine torque changed from motoring up to almost 80 % of the load according to ETC demanded values. The engine's thermodynamics, stability and good general behavior during the test demonstrate again the feasibility of this technology in transient operation.

Figure 11b shows the diesel and NG dynamic fuel consumption during the same last 300 seconds of the ETC test mentioned before. The figure pointed out the fast and coordinated response of the two fuels injected by the gas throttle electric valve and the diesel EUI which are governed by dual ECU to match engine fuel requirement during transient operation.

The exploded pie graph shown in Fig. 11b shows that 52 % of the total dual-fuel energy calculated by equation (2) is supplied by NG. This result corroborates the average GER around 50 % previously commented in the methodology section of this work (Fig. 6a) and also indicates the capability of the system for improving user economy during freeway-like transient conditions above all in places where diesel and NG prices have a substantial difference.

## 5. Conclusions

An experimental investigation was carried out in order to better understand the transient behavior of an existing heavy-duty diesel engine for on-highway truck applications when it is converted to a HDDF engine by means of the use of homogeneous natural gas injection in the intake line before turbocharger and the impact of the conversion on particle matter and $CO_2$ emissions. The most interesting aspects are remarked below.

- The feasibility of dual-fuel technology, especially the use of Homogeneous Gas Charge Injection in the intake line, was demonstrated during transient operation. Two worldwide standard cycles were used and in both of them dynamic behavior and transient engine response were matched to the original diesel baseline.
- The potential for PM specific emissions reduction working in dual-fuel mode was corroborated in stationary and transient engine operation. This result shows the capability of such engines to achieve a considerable decrement of one of the most current toxic contaminants in diesel engines.
- A decrement in $CO_2$ specific emissions in transient operation was also observed in dual-fuel mode compared with diesel only, which can contribute to diminishing the greenhouse emissions produced by this type of gas.
- The main drawbacks of this kind of technology found in this study were a considerable increment in NMHC specific emission and a small decrement of brake fuel conversion efficiency.
- $NO_x$ specific emissions in dual-fuel mode were increased during transient operation mainly due to EGR deactivation strategy. This result is not expected for non-EGR HD diesel engines with or without SCR system.

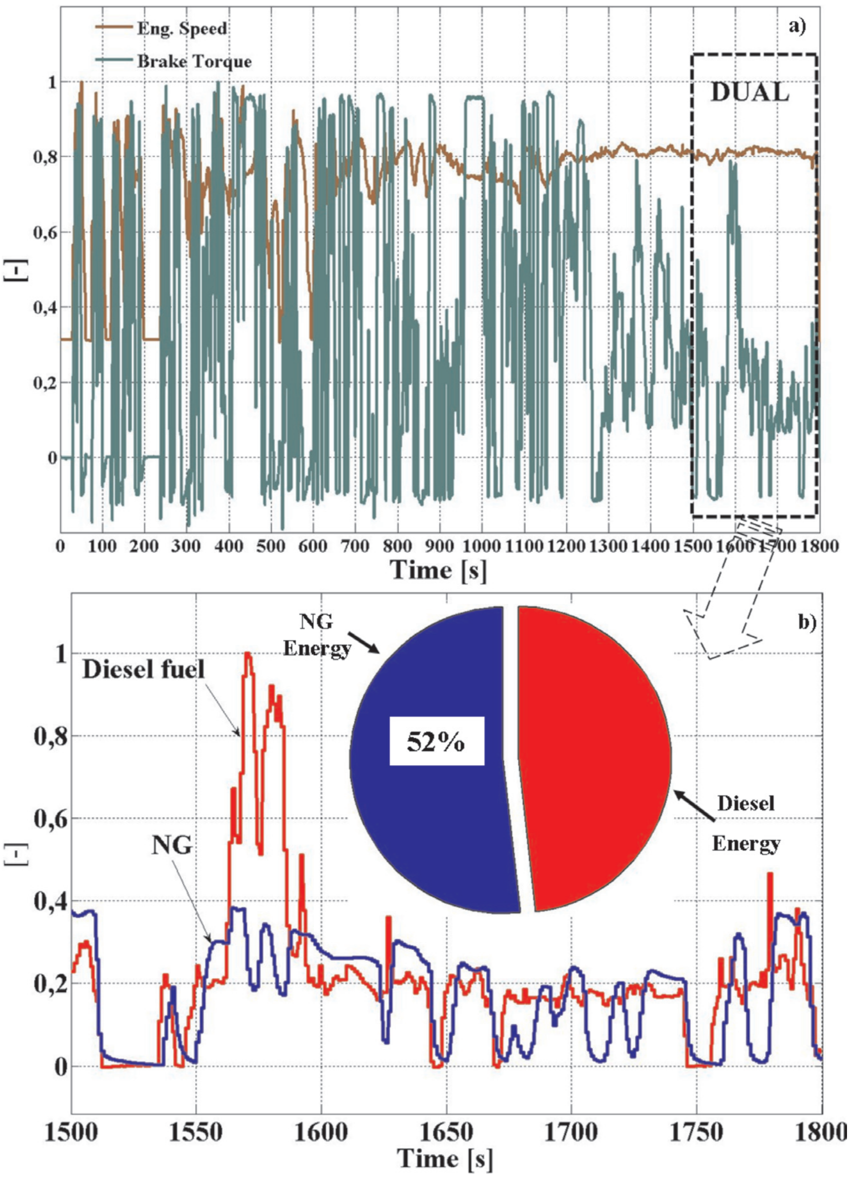



Fig. 11. ETC test: a) dual-fuel mode (motorway), b) NG and diesel energy distribution

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## Nomenclature

| | |
|---|---|
| CAN | Controller Area Network |
| $CH_4$ | Methane |
| C/H | Carbon Hydrogen ratio |
| CNG | Compressed Natural Gas |
| CO | Carbon monoxide |
| $CO_2$ | Carbon dioxide |
| DI | Direct Injection |
| ECU | Electronic Control Unit |
| EGR | Exhaust Gas Recirculation |
| EUI | Electronic Unit Injector |
| GHG | Greenhouse Gas |
| GUI | Graphical User Interface |
| HC | Unburned Hydrocarbons |
| HD | Heavy-Duty |
| HiL | Hardware in the Loop |
| LNG | Liquefied Natural Gas |
| LPG | Liquefied Petroleum Gas |
| NG | Natural Gas |
| NMHC | Non-Methane Unburned Hydrocarbons |
| $NO_x$ | Nitrogen Oxides |
| OEM | Original Equipment Manufacturer |
| PM | Particle Matter |
| RON | Research Octane Number |
| SCR | Selective Catalytic Reduction |
| U.S. | United States |
| VGT | Variable Geometric Turbine |

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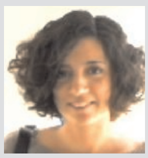

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